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A GRAPHICAL SOLUTION TO THE
GENERALIZED PARALLEL PUMP PROBLEM

By

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MASTER

A. TITLE:

A Graphical Solution to the Generalized Parallel Pump Problem

B. INTRODUCTION:

The usual method of combining pumps in system design, using pump head versus GPM curves, is not applicable if the pumps see different static heads and different dynamic losses. The designer then turns to iterative procedures centered on matching each parallel pump discharge line pressure to the common junction pressure. A faster and generalized method, not in universal use, is presented in this article with its technical basis. If not implementing it already, Damage Control Officer's should be made aware of this technique because it eliminates iteration and has a utility to address any pump configuration or requirement. The method is graphical and may be extended to multiple parallel pump installations.

C. BODY:

Most parallel pump installations are fitted with identical pump units for the system functional requirement, part interchangeability, and the uniform maintenance advantage. Call this type installation a Case 1 installation (see Figure 1) and note that since the pumps sit on the same pad, each pump has the same suction condition and the same discharge condition by design. It is not a revelation then that each pump also has the same system resistance curve due to friction losses, fitting losses, valves etc.

All engineers know how to combine Case 1 parallel pumps, but to refresh your memory and in the interest of a complete presentation, refer to Figure 2. On the same graph is plotted the vendor pump characteristic (Head vs. GPM), the pump suction static head, and the system resistance curve. Add P1 and P2 characteristic curves horizontally to arrive at the (P1+P2) curve. Draw a

horizontal dashed line to represent the pump suction head. Add to the suction head the system resistance at various flowrates and plot a (R+H) curve. The intersection of the (R+H) and the (P1+P2) curve is the operating point of the parallel pump installation. Drop down to the abscissa axis to read the combined pump flowrate (point A). Move horizontally to the ordinate axis to read the combined pump head (point B). That is all there is to a Case 1 paralleling problem.

The change made for Case 2 (see Figure 3) is the substitution of either pump P1 or P2 with a pump of different size. A little deeper theoretical basis will be developed to show that a parallel pump combination technique, similar to Figure 2, is also appropriate for Case 2.

Let Section 1 and Section 2 be defined as in Figure 3. The steady state energy equation (Reference 2) may be written:

$$\frac{p_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \left[\frac{p_2}{\rho g} + \frac{V_2^2}{2g} + z_2 \right] + h_s + h_L \dots \dots \dots \text{eq. (1)}$$

- where
- P = density, lbs./ft.³
 - g = acceleration of gravity, 32.17 ft./sec.²
 - p = pressure, psia
 - V = average velocity, ft./sec.
 - z = static head, ft.
 - h_s = delivered shaft head, ft.
 - h_L = head loss, ft.

Work done by the fluid is positive work, so for a pump (which does work on the fluid), h_s is negative. The head loss, h_L , is composed of friction losses, minor losses (such as fittings), and equipment losses. It is common practice to express minor losses in equivalent feet of straight pipe in order to apply the Darcy equation (equation 2). A Darcy equation is required for each line size.

$$h = f \frac{L}{D} \frac{V^2}{2g} \dots \dots \dots \text{eq. (2)}$$

where h = head loss due to friction/fittings, ft.
 f = Moody Chart friction factor
 L = Length of pipe, ft.
 D = Pipe inside diameter, ft.

The Moody Chart, correlating friction factor with relative roughness and Reynolds Number, can be found in any fluid mechanics textbook. Losses not considered minor losses, such as, control valves, filters, and other equipment have pressure drops varying with flowrate. Vendor data on these pieces of equipment will provide the necessary information. Pressure drop in psi, can be converted to a head loss in feet, by multiplying psi by 144 and dividing by the fluid density.

Rearranging equation (1) to account for line size variances, equipment losses, and negative shaft head, results in equation (3).

$$\frac{p_1}{\rho g} + \frac{V_1^2}{2g} + Z_1 + h_s = \left[\frac{p_2}{\rho g} + \frac{V_2^2}{2g} + Z_2 \right] + \left[\sum f \frac{L}{D} \frac{V^2}{2g} + \Delta h_e \right] \dots \text{eq. (3)}$$

Here the symbol Σ indicates a sum of Darcy equations and Δh_e is the head loss in feet due to equipment. Notice the left side of equation (3) is the pump inlet condition plus pump head, and the right side is the discharge condition plus system losses.

Using equation (3) compare the Case 1 problem to Case 2 problem. Remember that Case 1 has two identical pumps in parallel on the same installation pad. Case 2 is similar except the pumps are different sizes. For Case 1 and Case 2, the inlet conditions are the same. The discharge conditions are the same. The equipment losses are the same. A possible difference might be the losses associated with the suction and discharge branches from each pump to the common header. However, in addition to the fact that these lines are always short runs, it is the practice to size them so the pressure drop has the same $\Delta p/100$ feet of pipe also. In other words, the losses difference is usually negligible. If this is not accurate, one can always use the Case 3 method. For this article assume applicability and the parallel pump solution is given as in Figure 4.

The significant change in development from Figure 2 to Figure 4 is that because the pumps are different sizes, they also have different characteristic curves. The establishment of a combined operating point, and the identification of the combined flowrate and total head, are similar to Case 1 as developed in Figure 2.

Suppose the problem is generalized. Assume it is desired to operate two pumps in parallel which are not the same size, not located on the same pad, and do not take suction from the same source (see Figure 5). It is my experience that designers do not approach this problem graphically, but

rather calculate the junction pressure that supports the outlet condition. Knowing the required junction pressure, an iteration is performed on each pump branch until the branch outlet pressure matches the junction pressure. One then knows the operating point for each pump.

There is however a faster graphical technique which I came across in 1966 (see Reference 1). To my surprise there is a broad lack of knowledge on this method even among graduate level mechanical engineers. Perhaps it is time to rediscover it.

First observe that the steady state energy equation (equation 3) cannot be applied to Case 3 because each pump sees a different inlet condition and different resistances. Fortunately, there is one circumstance when Case 3 degenerates into Case 2. When the static plus pressure head of pump P1 equals the static plus pressure head of pump P2, and, the losses of branch 1 equal the losses of branch 2, Case 2 equals Case 3. The negation of any differences from this ideal situation makes a graphical solution possible. It can be achieved by subtracting out branch static, friction and pressure head losses, from each pump characteristic curve before the pumps are graphically paralleled. By subtracting these parameters from the characteristic curves, the effect is to present for paralleling modified curves which have zero branch system losses. In other words, the branch losses are equal (i.e., both are zero). Due to this manipulation, a Case 3 problem becomes a Case 2 problem. Referring to Figures 6 and 7, proceed as follows and all becomes clear.

For Pump P1:

- 1) Plot the pump head versus GPM curve from the original vendor characteristic curve of P1.
- 2) From the suction point to the P1 discharge junction point, plot the system resistance versus flowrate curve due to friction, fittings, and equipment. Call this curve R1.
- 3) Add the static head H1, and the pressure head p1, to the R1 curve developed in step (2) to obtain a (R1+H1+p1) curve.
- 4) Subtract vertically the (R1+H1+p1) curve from the P1 characteristic curve plotted in step (1), to obtain the modified "P1" curve. The result is Figure 6.

For Pump P2:

- 5) Repeat step (1) through step (4) for pump P2 to obtain the modified "P2" curve. A second plot, similar to Figure 6 results.

Paralleling Pump P1 and P2:

- 6) Horizontally add modified pump curve "P1" to modified pump curve "P2" to obtain curve "P1+P2".
- 7) Plot the system resistance versus flowrate curve for the common discharge line to obtain curve R3.
- 8) Add the static head H3, and the pressure head p3, to the R3 curve developed in step (7) to obtain the (R3+H3+p3) curve.
- 9) Circle the intersection of the "P1+P2" curve with the (R3+H3+p3) curve. The pump operating points are keyed to this intersection.
- 10) Refer to Figure 7. Drop down from the step (9) intersection to get the combined pump capacity in GPM (point E).

- 11) Go left horizontally from the step (9) intersection to the "P1" curve intercept. Drop down from the intercept to obtain the capacity contribution from pump P1. Go vertically up from the intercept to the P1 curve, then left horizontally to the ordinate axis (point D). This is the total head operating point for pump P1.
- 12) Go left horizontally from the step (9) intersection to the "P2" curve intercept. Down drop from the intercept to obtain the capacity contribution from pump P2. Go vertically up from the intercept to the P2 curve, then left horizontally to the ordinate axis (point C). This is the total head operating point for pump P2.

This is the graphical solution to a Case 3 generalized parallel pump problem. It appears complicated but really isn't and occasional use brings out its benefits.

A restriction on applicability should be discussed. Visualize an extreme example where a 1-inch discharge line ties into a 20-inch discharge line. It is intuitive that the small line junction pressure does not equal the large line pressure since the small flowstream is injected into the large flowstream. Since the tie-in pressures at the junction are not equal, none of the cases mentioned above apply. It is not until thorough mixing causes a common junction pressure that the graphical methods are valid. A problem involving connection of discharge pipes with an area ratio of over say 4:1 should be checked by the iteration technique to ensure graphical method applicability.

D. CONCLUSION:

A graphical method for combining parallel pumps under general configurations has been presented. The method may be extended to paralleling more than two pumps. This is possible because a modified pump curve can itself represent an already reduced system of more than one pump.

A graphical method may be reversed. Given the required outlet condition, a pump for combination with a second existing pump may be sized to meet a particular design objective.

It is suggested naval engineers become familiar with these methods so they are available for use in design and damage control problems.

E. REFERENCES:

- 1) Ferwerda, G. G. J., "New Graphical Method Determines Operating Points of Paralleled Pumps", Heating, Piping & Air Conditioning, December 1966.
- 2) White, F. M., "Fluid Mechanics", McGraw Hill Inc., New York, 1986.

F. BIBLIOGRAPHY:

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Figure 1 - Case 1

Figure 3 - Case 2

Figure 2 - Case 1
(Two Paralleled Pumps, Same Size, Same Configuration)

Figure 4 - Case 2
(Two Paralleled Pumps, Difference Sizes, Same Confirmation)

Figure 5 - Case 3

Figure 6
Modified Characteristic Curve
 (Vertically Subtract $(R_1 + H_1 + p_1)$ from P_1 to get " P_1 " Curve)

Figure 7 - Case 3
(Two Paralleled Pumps, Different Sizes, Different Configuration)

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